# Hardware-in-the-loop Testing of On-Off Controllers in Semi-Active Suspension Systems

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# HARDWARE-IN-THE-LOOP TESTING OF ON-OFF CONTROLLERS IN SEMI-ACTIVE SUSPENSION SYSTEMS*

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## ABSTRACT

This paper presents an experimental validation of a proposed Frequency Estimation-Based (*FEB*) controller for semi-active suspensions by using a Hardware-in-the-Loop (*HiL*) platform of a Quarter of Vehicle (*QoV*) model. The *FEB* approach is compared with three commercial *On-Off* controllers that have shown good results in comfort and road holding: *Sky-Hook* (*SH*), *Groud-Hook* (*GH*) and *Mix-1-sensor* (*M1S*). The comparison was done under the same experimental tests; the standards ISO-2631 and BS-6841 are used to evaluate the comfort and the Root Mean Square (*RMS*) index to quantify the road holding. The *QoV* model belongs to a front-left corner of a pick-up truck; the used experimental Magneto-Rheological (*MR*) damper is not symmetric and only hast 2 manipulation states. Experimental results show that the *FEB* controller has the best comfort performance at low frequencies (outperforms the benchmark controllers at 11.2%); while, for road holding, the improvement is slight; however, *FEB* controller works better for both goals simultaneously. By analyzing the suspension deflection, the *FEB* controller reduces up to 32.8% of motion respect to the *GH* controller. Additionally, the manipulation of the *SH* and *GH* controllers have several changes of actuation that do not allow the stabilization of the force in its desirable value; while *FEB* controller has a soft actuation defined on bandwidths.

*Keywords*: semi-active suspension control, hardware-in-the-loop, quarter of vehicle, MR damper

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## 1. INTRODUCTION

The main function of an Automotive Suspension Control System (*ASCS*) is to provide comfort by isolating the vehicle motion from the road irregularities, and to ensure the road holding in any driving condition. Due to the highly non-linear behavior of a shock absorber, this device is the key element in the *ASCS*. According to the properties for dissipating/absorbing energy, the automotive suspensions can be classified as passive, semi-active and active.

Passive dampers are the most used in commercial vehicles, the damping properties are designed mechanically; however, the constant damping coefficient has many limitations for ensuring comfort and safety simultaneously. On the other hand, semi-active and active dampers regulate the damping force through an *ASCS*. Normally, active suspensions achieve better comfort performance because they exploit the full *map force-velocity*; however, they require an external power system and its implementation could result more expensive than semi-active suspension systems.

Thus, semi-active dampers have variable damping force with lower power requirements; seminal results in [1] show that a semi-active suspension can decrease until 52% of vertical sprung mass acceleration. Commercially, there are two types of semi-active dampers: mono and double-tube. The mono-tube dampers are the most used; they can be Magneto-Rheological (*MR*), Electro-Hydraulic (*EH*) or pneumatic devices. The main advantages of the *MR* damper are: 1) fast time response (20-40ms), 2) large range of damping force and, 3) long bandwidth of control. An *MR* damper is a highly non-linear component with dissipative capability; its damping coefficient depends on the supplied electric current used to manipulate a magnetic field.

During last decades, there are several proposals of active and semi-active suspension control systems, some of them based on non-linear control theory [2-5], robust control solutions [6-8], optimum controllers [9], heuristic approaches [10-11] and *On-Off* strategies [12-14]. They have shown good results in comfort and some of them include criteria of safety. However, in an experimental evaluation, it has proved that semi-active dampers, with up to 25 operating states, only activates two states for an average of 92% of the driving time [15], concluding that an *On-Off* controller offers enough potential benefit to evaluate the semi-active property; additionally, the *On-Off* strategies are efficient and more feasible (light on-line computation and model-free synthesis) for being implemented in comparison with complex controllers. The *On-Off* control law computes the two values of manipulation as hard/soft damping.

Classical *On-Off* controllers such as *Sky-Hook* (*SH*) [12] and *Ground-Hook* (*GH*) [13] techniques have been exhaustively validated experimentally; however, they are not multi-objective. On the other hand, the *Mix-1-sensor* (*M1S*) controller [14] offers the possibility of selecting an exclusive control goal (comfort or road holding). In the aforementioned controllers, the controller output is not the damper manipulation; thus, the control law could compromise the controller practical feasibility by using a mapping algorithm from controller output to manipulation units. Hence, the design of a feasible and multi-objective *ASCS* is still an opportunity area. Recently, a novel controller based on the frequency estimation of motion is proposed in [16], the *Frequency Estimation-Based* (*FEB*) controller showed an efficient response for comfort and road holding in simulations by decreasing the trade-off between them.

This paper presents an experimental validation of the *FEB* controller in terms of comfort and road holding by using a Hardware-in-the-Loop (*HiL*) platform of a Quarter of Vehicle (*QoV*) model, the work is an extension of [16]. The *HiL* platform allows the validation of electronic control units and their algorithms by receiving and sending analogical signals from a mechanical component [17]. A comparative analysis among the *SH*, *GH*, *M1S* and *FEB* control strategies under the same experimental tests is presented; the standards ISO-2631 and BS-6841 are used to evaluate the comfort and the Root Mean Square (*RMS*) index to quantify the road holding. The *QoV* model is a front-left corner of a commercial pick-up truck; the experimental *MR* damper, manufactured by BWI$^{TM}$, is not symmetric and only hast two manipulation states.

The outline of this paper is as follows: the next section presents briefly the control law of all *On-Off* control strategies used in this comparative analysis. Section 3 describes the *QoV* model, the experimental system and the design of experiments. The results are discussed in section 4. Finally, conclusions are presented in section 5.

## 2. ON-OFF CONTROLLERS IN SEMI-ACTIVE SUSPENSIONS

*On-Off* control strategies have been analyzed exhaustively in experimental tests, e.g. in *test-rigs* or *HiL*. They are considered free of models because only need limit conditions in the required measurements for being implemented; they are robust in the experimental phase. At difference to the *test-rig*, a *HiL* platform uses a model of concentrated parameters of a vehicle which is programmed into a microprocessor, i.e. the non-linear vehicle dynamics is ensured into the model. The microprocessor receives/sends signals to a hydraulic system that acts over the semi-active damper.

Based on their published experimental performance, the classical *SH* and *GH* controllers are used as benchmark in this work; additionally, the *M1S* control strategy offers interesting simulation results with the innovation of using only one sensor. The control laws of these approaches are briefly described.

*Sky-Hook* (*SH*). It minimizes the vertical acceleration of the sprung mass by connecting a virtual damper between the chassis and sky. The control law is given by:

$$C_{SH} = \begin{cases} C_{max} & if \quad \dot{z}_s(\dot{z}_{def}) \geq 0 \\ C_{min} & if \quad \dot{z}_s(\dot{z}_{def}) < 0 \end{cases} \tag{1}$$

*Ground-Hook* (*GH*). It reduces the dynamic *tire-road* forces for optimizing the road holding index. A fictitious damping element is proposed between the wheel and the ground. It is a concept similar to the *SH* controller. The control law is:

$$C_{GH} = \begin{cases} C_{max} & if \quad -\dot{z}_{us}(\dot{z}_{def}) \geq 0 \\ C_{min} & if \quad -\dot{z}_{us}(\dot{z}_{def}) < 0 \end{cases} \tag{2}$$

*Mix-1-sensor* (*M1S*). It is a control strategy that only demands one sensor in the control system of a *QoV*: an accelerometer on the sprung mass. The control law selects, at the end of each sampling, the maximum or minimum damping coefficient according to a cross-over frequency $\alpha$, i.e. the term $(\ddot{z}_s^2 - \alpha_s^2 \dot{z}_s)$ is used as an instantaneous approximation in the time domain of the frequency range; when the controller is oriented to comfort $\alpha$ must be the frequency of resonance of the sprung mass. The control law is defined as,

$$C_{M1S} = \begin{cases} C_{max} & if \quad (\ddot{z}_s^2 - \alpha^2 \dot{z}_s) \leq 0 \\ C_{min} & if \quad (\ddot{z}_s^2 - \alpha^2 \dot{z}_s) > 0 \end{cases} \tag{3}$$

The damping in these reference controllers is considered directly proportional to the limit values of electric current of the experimental *MR* damper, i.e. $C_{min}{\sim}0$ A means a soft *MR* suspension and $C_{max}{\sim}2.5$ A means a hard *MR* suspension.

On the other hand, the proposed *FEB* controller is multi-objective (comfort and road holding) and its output is directly the manipulation signal over the semi-active damper, i.e. electric current. Following is briefly described its design.

### 2.1 Design of the *FEB* control strategy

This control strategy is based on the idea of monitoring the excitation frequency in the suspension for applying the required electric current into the semi-active damper in order to achieve the desired control goal [16]. It is proposed to estimate the frequency of motion as:

$$\hat{f} = \sqrt{\frac{\left(\dot{z}_{def_1}^2+\dot{z}_{def_2}^2+\cdots+\dot{z}_{def_n}^2\right)}{\left(z_{def_1}^2+z_{def_2}^2+\cdots+z_{def_n}^2\right)\cdot 4\pi^2}} \tag{4}$$

where, $n$ is the number of samples to compute of the relative position and velocity. The manipulation is obtained by analyzing the frequency response of the controlled variables, i.e. a *look-up* table between frequency and electric current is defined in order to assure the desired performances for comfort and road holding.

Four bandwidths of control have been defined; the first two are related to comfort and the remaining two to road holding and suspension deflection. If the applied electric current is not the required in a specific bandwidth, the control goals are not achieved. This is the reason for having an accurate online estimation of the frequency. The electric current values selected in the *look-up* table, Table 1, are the optimum for ensuring the minimum gain in the load transfer, i.e. optimum for comfort and road holding by using the experimental *MR* damper of two states.

Tab. 1. Look-up table based on comfort and road holding simultaneously.

| Bandwidth | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| $\hat{f}(=)Hz$ | 0-2 | 2-6 | 6-12 | 12-15 |
| $I(=)A$ | 2.5 | 0 | 2.5 | 0 |

## 3. EXPERIMENTAL SETUP

The *HiL* configuration was implemented by simulating a *QoV* model which was programmed in an FPGA device embedded in a Compact Reconfigurable I/O (*cRIO*) system of NI$^{TM}$. The model was discretized by using a sampling frequency of 300 Hz.

The vertical dynamics of the *QoV* model is given by (5). The model parameters are: the sprung mass ($m_s$= 711 Kg), unsprung mass ($m_{us}$= 81.5Kg), spring stiffness coefficient ($k_s$= 42,230N/m) and stiffness coefficient of wheel tire ($k_t$= 295,200N/m), that correspond to a commercial full size pick-up truck (front-left corner).

$$m_s\ddot{z}_s = -k_s(z_s - z_{us}) - F_{MR} \text{ and } m_{us}\ddot{z}_{us} = k_s(z_s - z_{us}) - k_t(z_{us} - z_r) + F_{MR} \tag{5}$$

where $\ddot{z}_s$ and $\ddot{z}_{us}$ represent the vertical accelerations of $m_s$ and $m_{us}$; $z_s$ and $z_{us}$ are the vertical positions and $z_r$ is the simulated road profile. $F_{MR}$ is the experimental *MR* damping force given by: $F_{MR} = F_d/\sqrt{R_d}$, where $F_d$ is the measurement of force obtained from the load cell and $R_d$ is the motion ratio (i.e. ratio between the vertical position of the damper experimentally and its position in a McPherson suspension).

Thus, in the *HiL* configuration, the suspension deflection generated by the simulation of the *QoV* is the input of a hydraulic actuator *MTS*$^{TM}$ in order to move the *MR* damper; the suspension deflection is limited for the damper stroke in its effects jounce/rebound (±50 mm). The measured *MR* force represents the *QoV* input and it is considered linearly distributed for the deflection between the masses. The *cRIO* monitors all process variables in a Human-Machine Interface (*HMI*).

Figure 1 shows the experimental system which is composed by 5 sections: (1) an operation and monitoring system by using an *HMI* designed in LabVIEW®, (2) the *MR* damper which has two states of actuation (0 and 2.5 A) and an impedance of 2 Ω, (3) an hydraulic system of actuation *MTS*™ 407 for controlling the damper rod position with capacity of 20,690 kPa and bandwidth of 15 Hz, (4) an electric current controller HCT™ that uses a *PWM* signal of 2 kHz from 0 to 10V and, (5) the simulator of a *QoV* model in real-time and Electronic Control Unit (*ECU*) by using a *cRIO* system.

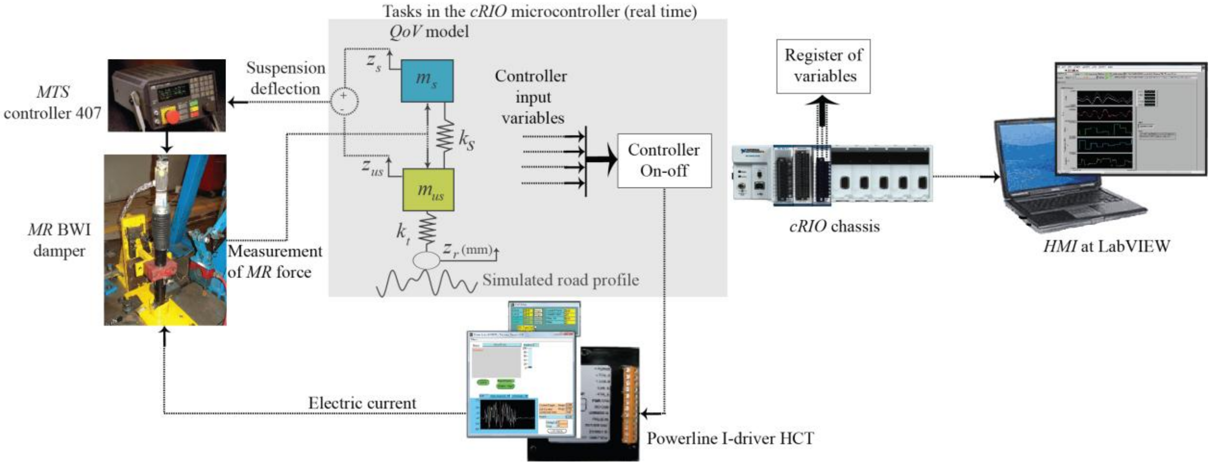


Fig 1. Experimental setup of the *HiL* configuration.

Two different tests of road profile over all control strategies were implemented in the *HiL* system. The test 1 is a road type F (rough road) according to the standard ISO-8606:1995, Fig. 2(a); this signal allows the analysis of the suspension motion in normal driving conditions. The test 2 represents a *Chirp* signal (0.25-5Hz) with decreasing amplitude (15-1mm), Fig. 2(b); this signal allows to explore uniformly the resonance motions of the sprung mass (comfort) and unsprung mass (road holding).

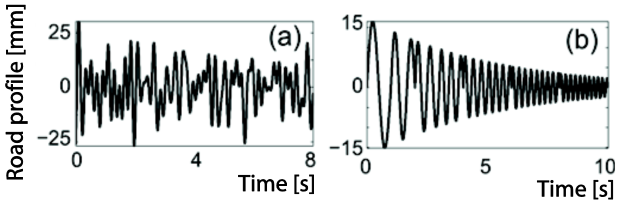


Fig 2. Implemented road profiles in the *HiL* configuration.

## 4. RESULTS

Experimental results of the *chirp* test are presented in Fig. 3 and 4. Figure 3 (a) shows that the *FEB* controller has the lowest gain in the vertical acceleration at low frequencies (<2Hz); after 2Hz, practically all suspension systems have same performance. For road holding, Fig. 3(b), *FEB* controller has acceptable performance, slightly better to the *GH* controller; however, the soft *MR* suspension (at 0A) presents the best road holding index at low frequencies. The difference between the *FEB* controller and the soft *MR* suspension is around 3 mm. Additionally, Fig. 3 shows the trade-off between comfort and road holding at low frequencies, i.e. the hard *MR* suspension (2.5 A) is good for comfort but limited for road holding and vice verse.

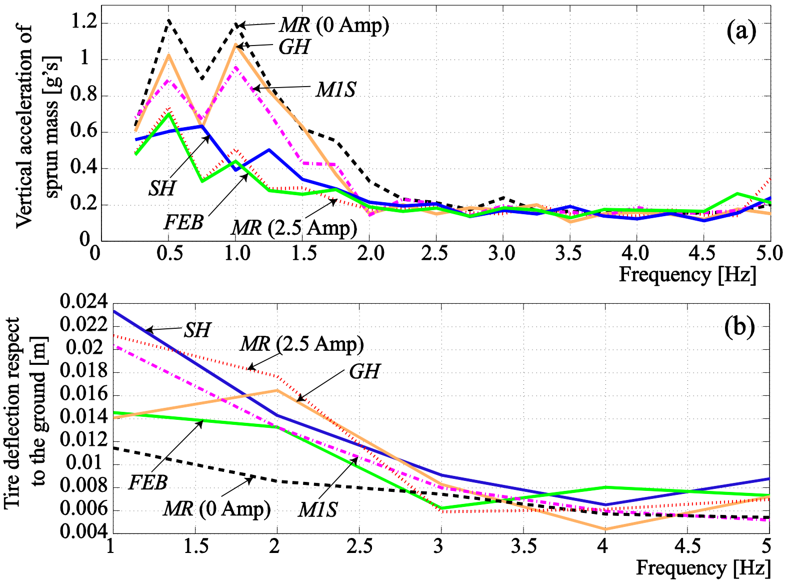


Fig 3. Analysis of comfort (up) and road holding (bottom) in the frequency domain (pseudo-Bode), by considering a *chirp* test as road profile in the *HiL* configuration.

The transient response of $\ddot{z}_s$, Fig. 4(a), shows that the magnitude is decreased up to 1 g whit the *FEB* controller in comparison with *GH* and *M1S*; this improvement is presented close to the resonance frequency of $m_s$ (~1Hz). By design of the *chirp* test, at beginning more suspension deflection is demanded, Fig. 4(b); however, *FEB* and *SH* do not show saturation in the displacement of the damper rod.

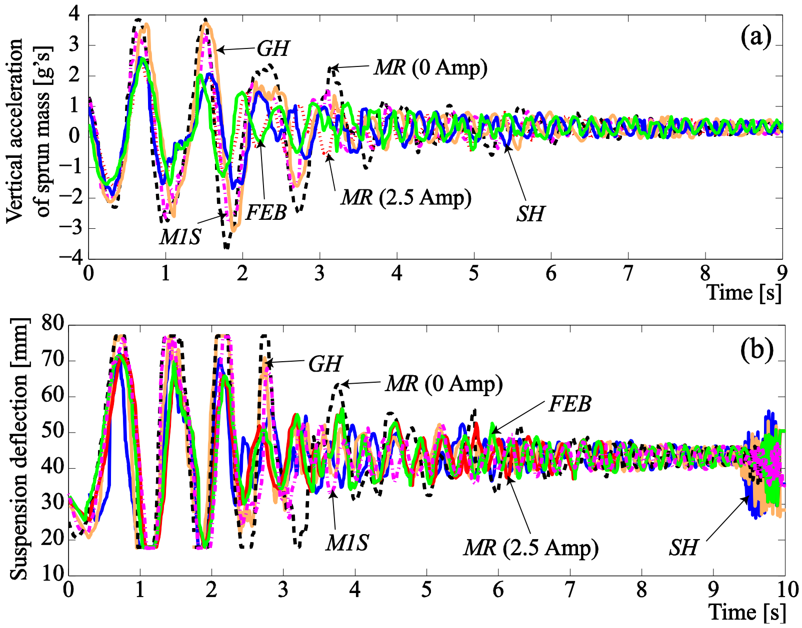


Fig. 4. Analysis of comfort (up) and suspension deflection (bottom) in the time domain, by considering the *chirp* test as road profile in the *HiL* configuration.

Table 2 shows the quantitative performance of comfort in both tests by using the standard BS 6841. In the *chirp* test, the *FEB* controller has the best performance i.e. an improvement between 11.2% and 39.4%, depending of the benchmark controller; practically few perception of movement according to the ISO 2631. For the ISO road test, *M1S* and *FEB* have the better comfort index. By analyzing the road holding in both tests, *RMS* index, there is not a significant difference among the controllers. Finally, the *FEB* controller showed the best performance in the suspension deflection for the *chirp* test (reduction between 7.2% and 32.8%); while, for the ISO road test, all controllers presented similar results.

Tab. 2. Analysis of comfort, road holding and suspension deflection in both tests.

| Controller | Maximum filtered acceleration by the standard BS6841 [m/s$^2$] | | *RMS* of the road holding index [cm] | | *RMS* of the suspension deflection index [mm] | |
|---|---|---|---|---|---|---|
| | *Chirp* test | ISO road test | *Chirp* test | ISO road test | *Chirp* test | ISO road test |
| *SH* | 0.107 | 0.190 | **0.7** | 1.4 | 9.7 | **2.8** |
| *GH* | 0.156 | 0.170 | **0.7** | 1.4 | 13.4 | **2.8** |
| *M1S* | 0.157 | **0.161** | 0.8 | 1.4 | 12.0 | 3.0 |
| *FEB* | **0.095** | 0.165 | 0.8 | **1.3** | **9.0** | 2.9 |

Figure 5 shows the manipulation of all controllers in the ISO road test. Clearly, the *SH* and *GH* controllers have several changes in the level of actuation caused by the sign of the deflection velocity in the semi-active operating zone; in some cases, these sudden changes do not allow the stabilization in the force signal in its desirable value. The *FEB* strategy did not present this problem because the manipulation is defined by bandwidths.

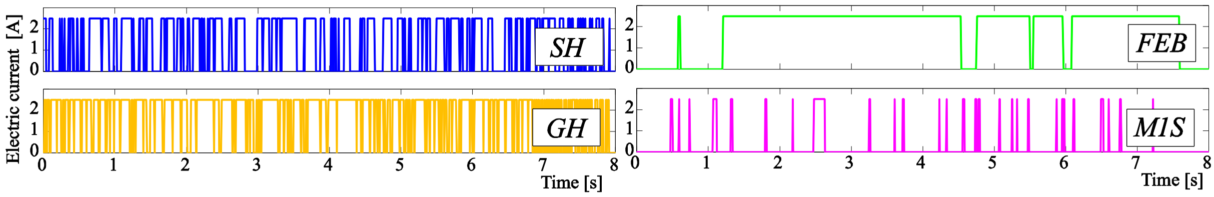


Fig. 5. Controller output in the control strategies for the ISO road test.

## 5. CONCLUSIONS

The *Frequency-Estimation-Based* (*FEB*) controller, which regulates the damping level based on an estimated frequency of motion, was experimentally validated by using a *Hardware-in-the-Loop* (*HiL*) configuration. The *FEB* controller was compared versus three commercial control strategies of semi-active suspension systems: *Sky-Hook*, *Ground-Hook* and *Mix-1-Sensor*. Experimental tests and standard monitoring indexes were designed in this comparative analysis for analyzing comfort and road holding. The *FEB* controller improved a 25% of comfort in average respect to the benchmark controllers and equaled the road holding performance. The controller design free of model, an easy computing, the generation of a soft continuous manipulation and, the demand of few measurements (low instrumentation cost) are advantages that make attractive this controller for comfort and road holding in an automotive semi-active suspension system.

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